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BUCKLING OF STEEL CYLINDERS CONTAINING CIRCULAR CUTOUTS REINFORCED ACCORDING TO THE AREA REPLACEMENT METHOD

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Abstract:

The effect of the use of the area replacement method (ARM) for reinforcing circular penetrations in cylindrical steel shells has been studied both experimentally and analytically. How this type of reinforcement affects the buckling strength of a shell subjected to uniform axial compression is the specific area of investigation. In shells that are of such a quality that the penetration reduces the buckling strength, the use of the ARM will increase the buckling strength of the shell. In any case, the conservative "knockdown" factors suggested for buckling design by the American Society of Mechanical Engineer's (ASME) Boiler and Pressure Vessel Code should ensure an adequate margin to failure under this loading condition.

Introduction:

Steel containments currently in use in reactor plant designs are basically large pressure vessels that consist of cylindrical sections in combination with a dome head (Fig. 1). The cylindrical section of the vessel has numerous penetrations, usually circular, for piping, personnel, and equipment access. Because of a number of loadings that will produce compressive states of stress in the walls of the vessel, the design must be examined for potential buckling. There are several questions that arise regarding the method or methods used to design these containments but the specific question addressed in this research concerns the use of the ASME Boiler and Pressure Vessel Code [1] rules for reinforcing around the penetrations. This code covers the reinforcing requirements for openings in shells and formed heads that is commonly known as the ARM. The purpose of this study was to investigate the use of the ARM for "fabricated" steel cylinders loaded in axial compression and having large circular openings. The radius-to-thickness ratio is in the range of that used for a nuclear power plant steel containment. A "fabricated" cylinder is here defined as a cylinder constructed by normal rolling and welding shop practices to normally specified engineering tolerances.

Reported buckling research on shells containing cutouts is summarized in Ref. 2. In addition, Miller and Grove [2] describe their own experiments on Mylar plastic cylindrical shells with reinforced circular openings. Babcock [3] has also conducted buckling tests on axially loaded right circular cylinders constructed of Mylar plastic, and Starnes [4] constructed and tested 16 right circular cylinders made of Mylar. Almroth and Holmes [5] tested aluminum cylinders machined from aluminum tubing; rectangular cutouts, with and without reinforcement, were evaluated.

It is important to note that:

1) None of these tests were conducted on "fabricated" steel cylinders as defined in the preceding section.

2) In these tests, each cylinder that was tested with a cutout was first tested before the cutout was made to establish a reference buckling load for that particular cylinder. This was possible since Mylar can be buckled many times with only negligible damage. The buckling deformation of the aluminum cylinders used by Almroth and Holmes was limited by a mandrel so that they could also be repeatedly buckled.

With the background given above, we are now in a position to give a statement of the problem specifically investigated in this research. When a fabricated steel shell structure is subjected to an axial load the buckling strength is affected by numerous factors, but the specific factors under investigation in this research are the area removed by a cutout and the amount that the cutout is reinforced. The ASME code specifies how the removed area (the cutout) is to be replaced (i.e., how the cutout is to be reinforced) so that the strength of a pressure vessel will be undiminished. The question to be addressed is: "will use of this ASME-ARM in a shell containing a cutout and subjected to an axial compressive load ensure that the buckling strength is also undiminished?"

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Description of the Test Cylinders:

Thirteen right circular cylinders were constructed of A321 stainless steel sheet as shown in Fig. 2. All cylinders had a nominal radius (R) to thickness (t) ratio (R/t) of 460, and a height (h) to diameter (D) ratio (h/D) of 1.09. Three of the cylinders contained no cutout and were used to establish the buckling load, and, thus, the "knockdown" factor for the fabricated shells constructed by the rolling and seam welding technique used in this study. Ten of the cylinders contained circular cutouts made by prepunching the opening at the cylinder midplane before rolling and seam welding. The ten cylinders that contained a cutout had a nominal cutout radius (r) to $\sqrt{Rt}$ ratio of $r/\sqrt{Rt} = 3.64$. This dimensionless ratio is widely used to characterize cutout size. Eight of the cylinders containing cutouts were reinforced according to the ASME-ARM procedure. The amount of reinforcement [Area replaced (A_r) to area removed (A)], A_r/A varied between 0 and 100%.

Comparisons of the various dimensions and ratios of the cylinders used in this study to those used in previous research by others are given in Table I. Comparison is also made to the Watts Bar Nuclear Plant (WBNP) containment. The "cutout" used from the WBNP data is the large equipment hatch, typically the largest opening in a containment shell.

TABLE I
RELATIVE SHELL SIZES

Material	Los Alamos Tests A321 SS	Miller (Ref. 2) Mylar[2]	Babcock (Ref. 3) Mylar[2]	Starnes (Ref. 4) Mylar[3]	Almroth (Ref. 5) 6061-T6 Al Tube[4]	WBNP Steel
R/t	460	568	417	400 533 800	430 675	460
$\bar{r} = r/\sqrt{Rt}$	3.64	6.36	1.28	0 - 6	Note #5	3.73
h/D	1.09	1.00	0.975	1.25	1.00	1.00
A_r/A	0 - 100%	0 - 100%	Note #1	Note #1	Note #1	----

1. These tests did not involve ASME-ARM reinforcement.
2. A single cylinder, modified and retested
3. 16 cylinders, each one retested after modification.
4. 11 cylinders, each one retested after modification.
5. Rectangular cutouts.

Data were taken on the test cylinders to characterize the magnitude and form of the geometrical imperfections introduced by the fabricating process used to construct the test cylinders. These measurements are discussed in Ref. 6.

Experimental Studies:

The cylinders used in these tests were instrumented using bonded resistance strain gages located as shown in Fig. 3. These gages were installed to determine the distribution of strain and, by inference, the magnitude of the membrane and local bending stresses.

The cylinder to be tested was mounted in a 222,400 N (50,000 lb) MTS hydraulic testing machine. Thirteen-mm-(0.5 in.)-thick teflon pads were placed between the cylinder ends and the testing machine self-aligning heads. The typical test procedure consisted of loading the cylinder to approximately 25% of the expected buckling load while monitoring the axial membrane strain around the top of the cylinder. If necessary, the cylinder was then unloaded and rotated and/or shimmed to give a more uniform distribution of load. This adjustment of load was repeated as necessary until the membrane strain at the top of the cylinder was as uniform as possible. The cylinder was then loaded to, and beyond, the buckling point using a cross-head loading rate of 0.50 mm (0.100 in.) per minute. The strain gages, the cross-head position, and the load cell were monitored and recorded continuously with the data being fed into an on-line computerized data reduction system.

Table II summarizes the results of the experiments. In the literature, it is customary to report "knockdown factors" or "capacity reduction factors" (α) as the ratio of the measured buckling load (P) to the classical buckling load (P_{cl}), that is, $\alpha = P/P_{cl}$. The classical buckling load for cylinders used in these tests under a uniform axial load is given by

$$P_{cl} = \frac{2\pi E t^2}{\sqrt{3(1-\nu^2)}} = 203.0 \text{ kN } (45{,}630 \text{ lb}),$$

where E = Young's modulus of elasticity = 206.8 GPa (30×10^6 psi);
t = wall thickness = 0.508 mm (0.020 in); and
ν = Poisson's ratio = 0.3.

Figure 4 is taken from Ref. 7 and is a plot of the capacity reduction factor for use in the design of fabricated shells with no cutouts when the length parameter $M \equiv L/\sqrt{Rt}$ is greater than 10. Because the ratio $L/\sqrt{Rt}$ for the cylinders used in this study is 42, this curve applies. The ratios of P/P_{cl} for the three test cylinders without cutouts and their average values are shown on this figure.

In Table II we have also reported values of "first detectable buckle." This value is the load for which the shell exhibited the first indication of impending failure by structural instability. In nearly all tests this load was accompanied by the buckle appearing in the shell wall as an elastic "snap through," detectable by both sound and a "jump" in several of the strain gage records. Configurations after the first detectable buckle were always stable and generally barely detectable in the load versus cross-head displacement curves. Since these first detectable buckles are local in nature they may be highly dependent on the magnitude of the local imperfections.

The data obtained from the test cylinders containing cutouts (tests 4-13 in Table II) have been plotted in two ways. First, the average buckling load for the three cylinders without cutouts (tests 1-3, Table II) is computed. This value, P_0, is found to be 49.82 kN (11,200 lb). The buckling load of each cylinder with a cutout is then divided by P_0 to obtain the ratio of P/P_0. Figure 5 shows the ratio of P/P_0 plotted vs the percentage of reinforcement together with data taken from Ref. 2. The data from the present tests do not fit the trend established by the data from Ref. 2 and, hence, do not directly support the conclusion that when a cutout is reinforced with A_r/A of 100%, the buckling strength is at least as great as that of a cylinder without a cutout.

The apparent discrepancy between the results of the present tests and the data taken from Ref. 2 can be attributed to cylinder quality, method of obtaining P_0, and the actual load distribution applied to the cylinder. The first two effects are discussed in the remainder of this section. Effects of load distribution are discussed in detail in Ref. 6.

The discrepancy can be partially explained by plotting the ratio of actual buckling load for each cylinder to the classical value, P_{cl} (see Fig. 6). In this figure, reinforcement is not being considered. Hence, only data from 'no cutout' tests ($\bar{r} = 0$) and unreinforced cutout tests ($A_r/A = 0$) are considered. The solid line curves are reproduced from Ref. 4. As Starnes[4] points out, with high quality shells ($P/P_{cl} \geq 60\%$ for cylinders used in his tests) a small hole (say $\bar{r} \leq 0.5$) has no effect on buckling load because the effect of the hole "is apparently too small to cause buckling before the shell buckles into the general collapse mode due to some other imperfection." Progressively larger holes cause the buckling load to progressively decrease as shown in Fig. 6.

The fabricated shells used in the present tests were of poor quality. For the three shells tested without cutouts, the values of P/P_{cl} were 21.6, 25.6, and 26.4%. However, as shown in Fig. 4, these are the values to be expected in fabricated shells, and as shown in Fig. 6, the data from the present tests also support the speculation discussed above. Specifically, with a cutout size such that $r/\sqrt{Rt} = 3.64$, it is not clear that even the unreinforced cutout significantly changes the buckling load. See data points (■) on Fig. 6.

Supporting Analyses:

A finite element computer code has been applied to the same problem to support the experimental work that has been described. A complete discussion of this analysis is given in Ref. 6. Results are summarized here. First, the code predicts a critical buckling load of 97% of P_{cl}, that is, $P/P_{cl} = 0.97$, and the number of buckling waves were in agreement with the classical solution. Next, the code was run for a condition of uniform axial loading to examine the effect of the hole. As expected, the buckling waves formed around the hole and the critical load was reduced to 15% of that for a cylinder containing no hole. Thus, the analytical solution predicts that the cutout causes a much greater reduction in buckling load than is predicted by any of the experiments. This is in accordance with the speculation, mentioned in the preceding section, that the more nearly "perfect" the virgin cylinder the greater the effect of the cutout.

The analytical model was then modified to simulate 100% reinforcement, applied according to ASME-ARM. The applied load was uniformly distributed. For this case, the buckling load was found to be 74% of P_{cl} (i.e., $P/P_{cl} = 0.74$), and the buckling began around the hole. Thus, the analytical solution predicts that 100% ARM reinforcement greatly increases the buckling strength of a cylinder containing a cutout, but it fails to confirm that 100% ARM reinforcement will restore the buckling strength to the value for the "no cutout" condition. Because the cylinder still buckled around the hole, increasing the buckling load to or above the classical value of a perfect cylinder would probably require not only more reinforcing, but also that it be spread further away from the hole than is allowed by the ARM method.

Finally, the analytical model was used to investigate the effect of nonuniform loading. A study of the strain readings obtained from the gages located near the top of each test cylinder revealed that the load was, in most cases, not uniformly distributed around the circumference. The analytical model with an unreinforced cutout was loaded to simulate the extremes in load distribution indicated by the strain gages.

TABLE II
SUMMARY OF EXPERIMENTAL RESULTS

Test No.[1]	Cylinder Description[2]	Reinforcement (A_r/A in %)	Buckling Load 1st Buckle kN (lb)	Buckling Load Collapse kN (lb)	P/P_{c1} % Note #3	P/P_o % Note #4
1[6]	#1--No cutout	--	36.9 (8300)	53.7 (12,070)	26.4	
2	#2--No cutout	--	32.9 (7400)	43.8 (9840)	21.6	
3	#3--No cutout	--	37.4 (8400)	52.0 (11,690)	25.6	
4	#10--Cutout, no reinforcement	0	26.6 (5990)	39.7 (8925)	19.6	79.7
5	#11--Cutout, no reinforcement	0	37.6 (8450)	54.3 (12,200)	26.7	109
6	#4--Reinforced cutout	33	42.3 (9500)	46.2 (10,390)	22.8	92.8
7	#8--Reinforced cutout	33	26.7 (6000)	45.1 (10,150)	22.2	90.6
8	#5--Reinforced cutout	76	28.0 (6300)	48.6 (10,920)	23.9	97.5
9	#6--Reinforced cutout	81	53.4 (12,000)	61.6 (13,840)	30.3	124
10	#9--Reinforced cutout	102	Note #5			
11	#7--Reinforced cutout	102	53.4 (12,000)	59.5 (13,370)	29.3	119
12	#12--Reinforced cutout	52	33.4 (7500)	39.4 (8860)	19.4	79.1
13	#13--Reinforced cutout	101	31.6 (7100)	40.5 (9100)	19.9	81.3

NOTES:

1. In chronological order.

2. For all cylinders: $R/t = 460$, $h/2R = 1.09$
   For all cutouts: $\bar{r} = r/\sqrt{Rt} = 3.64$.

3. P = Collapse buckling load
   P_{c1} = Buckling load as computed by classical theory = 203 kN (45,630 lb).

4. P_o = Average collapse buckling load for the three cylinders without cutout = 49.[illegible] kN (11,200 lb).

5. Error in testing (goof); buckled at unknown load.

6. On the first test only, testing machine on load control. On all other tests, testing machine on stroke control of 0.5 mm/min.

With the load applied over the cutout increased to 26% above the average load, the predicted P/P_{c1} ratio is 12% (as compared to 15% with uniform loading). When the load applied over the cutout is reduced by 45% below average load, the predicted P/P_{c1} ratio is 28%.

All of the results of these analytical studies are summarized in Table III. The authors note that many buckling tests reported in the literature state that "the test cylinder was subject to a uniform load" but in only a very few cases is data offered to verify this statement. The reported buckling

TABLE III
SUMMARY OF ANALYTICAL STUDIES

Case	Buckling Load (lb)	(kN)	P/P_{cl}
Perfect cylinder--uniform load	50,020	(222.5)	0.97
Cylinder w/hole, $\bar{r}$ = 3.5 (no other imperfection)	7,634	(33.96)	0.153
Cylinder w/hole, $\bar{r}$ = 3.5, 100% reinforcement, (no other imperfection)	36,974	(164.5)	0.74
Cylinder w/hole, (Hole over loaded by 26%)	6,107	(27.16)	0.12
Cylinder w/hole, (Hole under loaded by 43%)	13,854	(61.62)	0.28

strengths range from $0.19 \leq P/P_{cl} \leq 0.60$ for R/t = 460, and, in most cases, imperfections are used to explain this wide variation in results. It is the authors' opinion, based on the preceding analysis, that nonuniform loading is also an important, and often unknown, contributor to the wide scatter in the experimental data.

Conclusions:

The exact values of the buckling load of a fabricated steel shell without a cutout may vary within rather wide limits. When a cutout is introduced into that shell, the effect of the cutout on the buckling load depends not only on the size of the cutout, but, also, on the buckling strength of the virgin shell. Furthermore, the effect that reinforcement of the cutout will have also depends upon the buckling strength of the virgin shell.

When a cutout is made in a fabricated steel shell of poor quality (low P/P_{cl} or low value of knockdown factor, presumably due to large initial imperfections), the buckling load may be reduced only slightly, or not at all, by the cutout, and reinforcement will have little or no effect. In this case, the margin to failure is ensured by the conservative knockdown factor required by the ASME code. Reinforcement of cutouts, according to the ASME-ARM, should ensure that if the buckling strength of the shell is not governed by initial imperfections the effect of the cutout will be reduced by the reinforcement and the margin to failure will be increased above the value ensured by the use of a conservative knockdown factor.

An investigation of the importance of loading conditions is needed to 1) better understand the so called 'simple' load cases used to investigate the importance of imperfections and to check theory, and 2) because variation in loading conditions is a certainty in the real world.

Acknowledgements:

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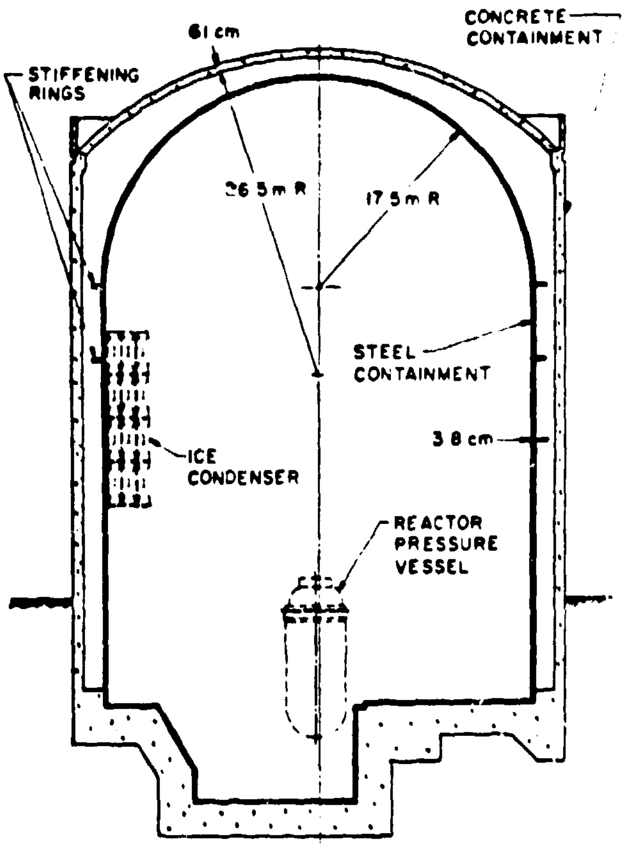

Fig. 1. Reactor containment.

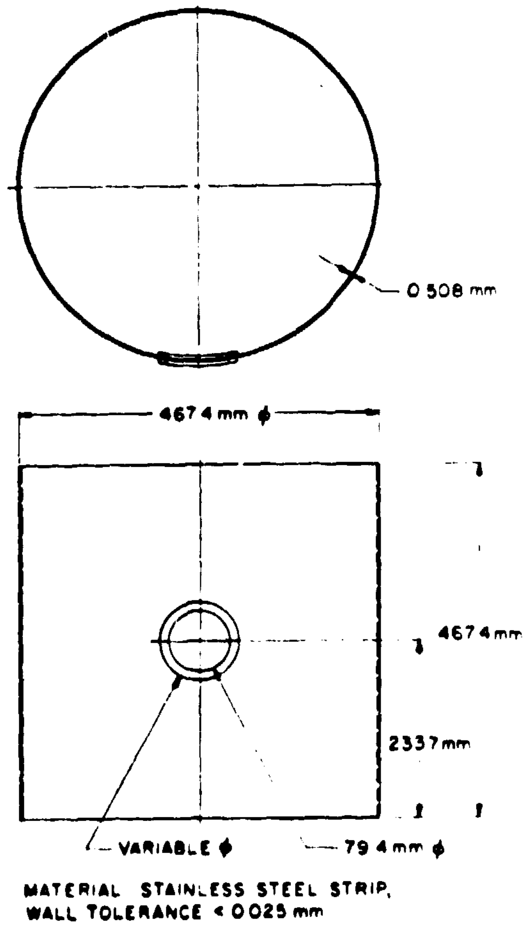

Fig. 2. Test specimen.

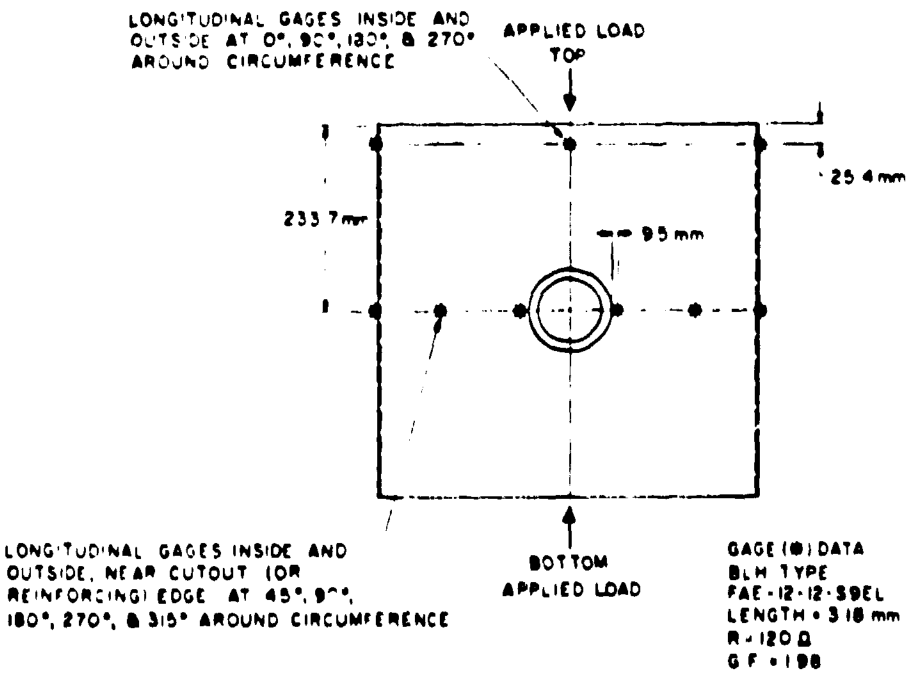

Fig. 3. Strain gage locations.

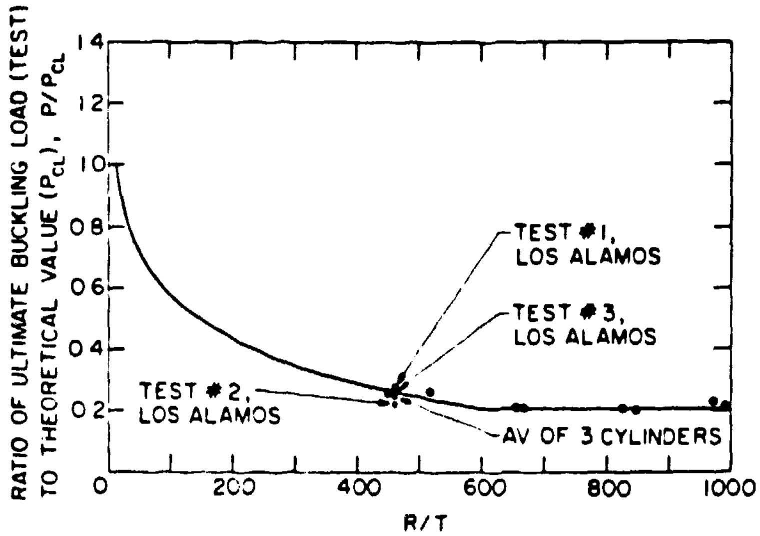

Fig. 4. Recommended capacity reduction factors for fabricated steel shells.

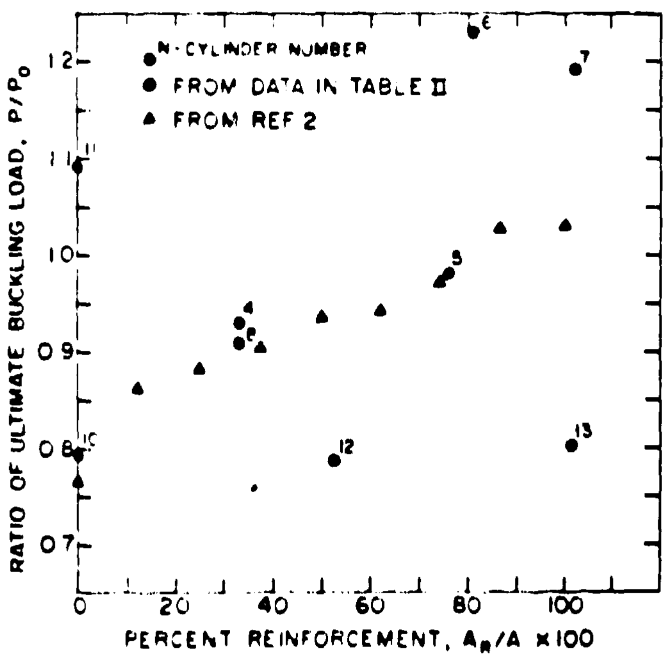

Fig. 5. Effect of reinforcement.

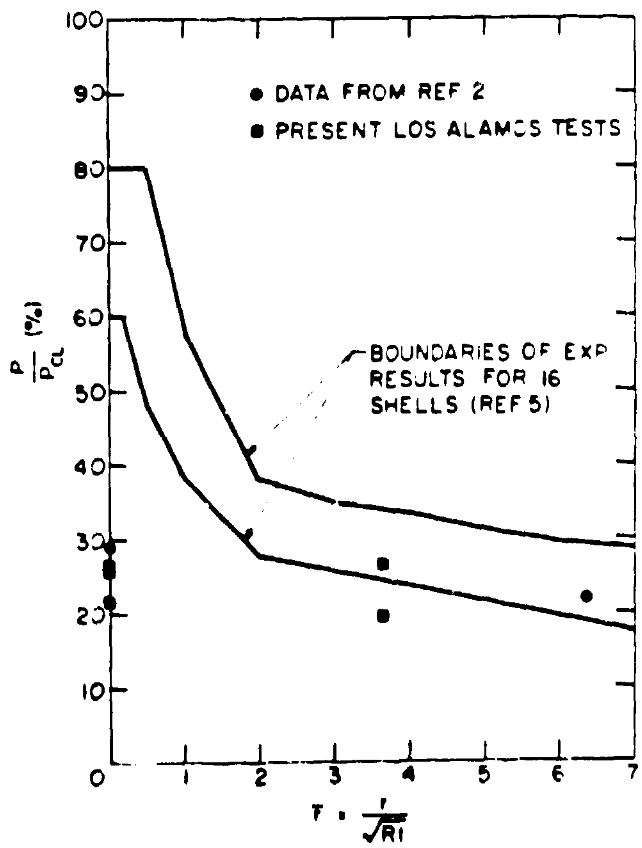

Fig. 6. Effect of cutout--no reinforcement.